Proceedings of ICAPP 2015
May 03-06, 2015 - Nice (France)
Paper 15259

Performances and Cost Assessment of an Air Cooled Two-Stage Rankine Cycle for Large Power Plants Operating with Different Working Fluids

Bo Liu[1], Franck David[1]*, Christophe Coquelet[2], Renaud Gicquel[3], Philippe Riviere[4]
[1]: EDF R&D, 6 quai Watier BP 49, 78401 Chatou Cedex, France
[2] Mines Paristech, PSL Research University, CTP- Centre for Thermodynamic of Processes, 35 rue St-Honoré – 77305 Fontainebleau, France
[3] Mines Paristech, PSL Research University, PERSEE, Centre for processes, renewable energy and energy systems, CS 10207 rue Claude Daunesse, 06904 Sophia Antipolis Cedex, France
[4] Mines Paristech, PSL Research University, CES, Centre for energy efficiency of systems, 60 Bvd St Michel, 75006 Paris, France
**: Corresponding author: E-mail: franck.David@edf.fr.*

Abstract *–Steam/water Rankine cycles are mainly used as Energy Conversion Systems (ECS) for large power plants. Due to the low density of steam at low pressure, the turbine size is generally very big. Furthermore water cooling is almost always preferred as cooling option since it provides the best energy efficiency as compared to air cooling. Nevertheless, water use can be strengthened in the future due to environmental constraints (withdrawal or heat release limitations).*

In the present work, a two-stage Rankine cycle (TSRC) is considered, combining a steam/water cycle and organic Rankine cycle (ORC) cooled by an air cooled condenser (ACC). The backpressure of the steam cycle is limited and the heat which remains in the steam is transferred to an ORC through a condenser-boiler.

Due to the organic fluid high density, it is possible to reduce the installation size. Furthermore, the organic fluid is likely to provide additional power when ambient temperatures are low. With those two advantages it is expected for dry cooling to become more cost effective.

The paper reports the methodology that has been used in order to optimize a design in terms of fluid selection, component sizing and efficiency. Preliminary cost analyses related to the different working fluids and to local ambient conditions are given.

I. INTRODUCTION

I.A. Energy Conversion Systems: Steam Water Rankine cycle and ORC

The steam/water Rankine cycle with water-cooling is the most widespread system for high temperature heat power production in nuclear power plant or large generation plant. An optimal energy efficiency with this technology is completed when the cycle operates with a turbine backpressure as low as possible. Nevertheless, it leads to several weaknesses that can be pointed out: particularly the important turbine size due to the low density of steam and its dependence on water withdrawal for direct cooling or to compensate evaporation in wet cooling towers. The use of water provides the better and most efficient technological ways to release power in the "cold" heat sink but can become difficult in the future with respects to environmental regulation or climate evolution.

An improvement can be made to reduce the turbine size and make using air-cooled condenser possible, by adopting a multi-staged cycle architecture: combining a classic steam Rankine cycle with an ORC which operates with organic fluid as working fluid instead of water.

The use of ORC has been proven to be a promising solution for low grade heat power production such as a biomass power plant [1] and geothermal power plant [2]. It is more suitable for low temperature heat recovery than water steam. The cycle performance and its cost effectiveness closely depend on the chosen working fluids, its architecture and the operating conditions [3]. In the standard ORC applications, the temperature conditions on the heat source are fixed and a temperature glide can be observed. Thus it is necessary to select the adequate fluid and cycle architecture in order to obtain the best performance.

Numbers of papers investigated ORC concepts in terms of fluid selection, cycle efficiencies and cost effectiveness for different applications and heat sources. Saleh et al. [4] made a screening of 31 pure working fluids and observed the following tendency: cycle thermal efficiency is generally better for fluids with high critical temperature. When considering an ideal ORC cycle and an isentropic fluid, Liu et al. [5] were able to express the thermal efficiency according to the critical temperature and conclude that the thermal efficiency for various working fluids is a weak function of the critical temperature. Maizza et al. [6] suggested using high latent heat and high density fluid in order to decrease the size of heat exchangers.

Different cycle architectures have also been studied for ORC applications. Liu et al. [7] studied several types of Organic Rankine Cycles with subcooling or superheating and concluded that superheating and subcooling are detrimental to ORC efficiency. Roy et al. [8] analyzed the performance of an ORC with superheating, operating with R12, R123, R134a and R717. The basic ORC can be modified by incorporating turbine bleeding or a regeneration process to improve its efficiency [9]. Desideri et al. [2] studied different types of Rankine cycles for geothermal heat recovery and concluded that regenerated Rankine cycle is a promising solution for the exploitation of low temperature liquid-dominated geothermal source.

However, no single fluid has been identified as optimal for ORC application due to the strong interdependence between the optimal fluid, the working condition and the cycle architecture [10].

I.B Main Features of a Two Stage Rankine Cycle

In our application, we focused on an energy conversion cycle which is slightly different to the traditional ORC application. The system is a modified classic subcritical steam Rankine cycle for electricity production. Instead of expanding the steam to extremely low pressure, the vapor leaves the turbine at a higher pressure hereafter called "Steam Turbine Exhaust Pressure (STEP)". The thermal energy available in the steam is transferred to an ORC through a condenser-reboiler for low grade heat recovery. Such a process is similar to bottoming cycles [11]. However, the heat source for the ORC is, in our case, controlled by the STEP since its temperature is set by the steam condensation temperature. The steam condensation temperature lies between 70 °C and 120 °C, corresponding to a STEP from 0.3 bar to 2 bar. Consequently, we are mostly interested in sub-critical ORC where the evaporation temperature is constant. The STEP appears as a cycle parameter involved in the efficiency optimization design in relation with the ORC fluid.

I.C. Motivations and Expected Advantages

Reduction of installation size and limitation of water consumption

On leaving the turbine at a higher pressure, water steam volume is much smaller. Therefore, it is possible to greatly reduce the size of the steam turbine. At the same time, density of organic fluid vapor in the second cycle is generally much higher than expanded water steam. The turbine is more compact. Thus we are able to largely reduce the size of the installation and its investment cost.

Furthermore, by using high density fluid, it is possible to transport and exchange large amount of energy through an air cooled condenser in order to use air as a cold source instead of water. Although this leads to a larger heat exchange surface areas than for conventional wet cooling process, it enables to eliminate the dependency on water for cooling.

The advantage of such a system is that it can be sited on a chosen site independent of the great and constant availability of a water source. The possibility can be valuable to comply with different objectives as it may offer more solutions to set the production plant according to transportation grid needs, or according to lower risk potentiality of natural disaster. In addition the system, if cost effective, can anticipate potential future restriction in water withdrawal and thus may facilitate long term operations.

Reduction of kinetic energy losses downstream of the turbine and increase of power production during cold ambient temperatures

Another advantage of such a two stage cycle using high density fluid is the possibility to reduce the leaving losses of the turbine and produce more electrical energy at lower cold source temperature. Actually, while the external temperature is low, the turbine should be able to produce more power. However, this gain is tempered by the leaving losses in the turbine. As the volume of water steam becomes very large at low pressure according to low ambient temperature, the kinetic energy increases and cannot be completely converted by the turbine. Operating the cycle with a higher density working fluid can reduce the leaving losses. Thus the system operates with better performance when the cold source temperature is low.

Finally, while having a more compact machine room and using air as cold source without any evaporation, the two stage cycle power plant is able to achieve increased performance at low temperature because of low turbine leaving velocity losses.

This concept of two stage cycle has been investigated by EDF (Électricité de France) in the 80s. A 22 MWe medium scale prototype was developed near Paris [12], [13]. In this early prototype, instead of using an organic fluid, ammonia was chosen as a working fluid for its high density and high latent heat. However, this fluid is toxic [14]. To prevent any risk for operators and environment, specific designs and operation procedures were developed. The objectives of this experimental study were to verify the system performance and master the technologies for the design and manufacture of the special cycle components. The cost of such a two-stage cycle architecture was studied by EDF for 1300 MWe nuclear power plant and it revealed this concept might be economically relevant as compared to a standard steam cycle [12]. The concept of multi-stage, using ammonia, for large steam and gas-steam power plants has also been discussed by Desideri et al. [15].

I.D. Details of the Study.

In the first section, we present the main criteria for the fluid selection in the framework of our study and plant design. In a second section, we present and discuss the characteristic of the parametric study of the cycle efficiency. A third part is dedicated to the assessment of main component designs, heat exchangers and turbines, for a limited list of selected fluids. Finally, by making "off-desin" cycle performance calculations, we have derived preliminary evaluations of cost effectiveness depending on fluid selected, for a location in France.

II. WORKING FLUID CRITERIA

The choice of a fluid to be included in the ORC should respect multiple criteria. In this section we present and discuss the main features of an ideal fluid in order to guaranty the cycle performance, to be adapted to the system operating conditions, to satisfy environment protection standards and to ensure operator's safety.

Therefore, the fluid selection criteria for our two-stage Rankine cycle are listed as below.

CRITICAL TEMPERATURE

In order to guaranty a better ORC efficiency and be able to run sub-critical cycles with high STEP, the critical temperature of the chosen fluid shall be as high as possible comparing to the evaporation temperature. In addition a numerical study of cycle efficiency has been carried out with different working fluids and will be discussed later.

SPECIFIC VOLUME AND LATENT HEAT

One main target of using two-stage Rankine cycle architecture is to reduce the installation size. To that existent, the latent heat of the chosen working fluid need to be as high as possible, the specific volume as low as possible.

TRANSPORT PROPERTIES (THERMAL CONDUCTIVITY AND VISCOSITY)

The transport properties are necessary to estimate the surfaces of heat exchangers. Particularly, the conductivity shall be as high as possible to reduce the exchange surface needed.

TRIPLE POINT TEMPERATURE

By using the air cooled condenser, we have to take into account that the temperature periodic variations of the cold source (ie dry air) are larger than for water. As a consequence, the condensation temperature of the organic fluid can become very low in winter. The fluid should not be freezing (in the ORC) under low temperature condition.

LOW PRESSURE IN THE ORC

Ideally, the condensation pressure of the fluid in the ORC should be slightly higher than the atmospheric air pressure. This would ensure no air enters the ORC installation.

GLOBAL WARMING POTENTIAL (GWP) AND OZONE DEPLETION POTENTIAL (ODP)

Environmental restriction has much hardened in the 2 last decades [16-17]. The 2014 European F-gas directive plans the prohibition of fluorinated greenhouse gases with a GWP of 2 500 or more from 2020. It will also target the fluorinated fluids with a GWP of 150 or higher in the medium term [29].

Thus, the choice of working fluid for the two-stage Rankine cycle should follow the current environment protection standards and its likely future trends. The fluid should have a null ODP and a GWP as low as possible in order to be accepted and to limit the risk of phase out decision, as it is experienced with HFC.

TOXICITY AND INFLAMMABILITY

For operators' safety, it is wise to prefer the use a non-toxic and non-inflammable organic fluid following the Ashrae standards classification [14]. However in the phase of investigation on the cycle efficiency, we do not eliminate any fluid first. In fact, as we investigate pure fluids in this paper, toxicity and inflammability may lead us to delete high performance fluid from the potential candidates. However, these fluids could be used in mixtures. For this reason, this criterion, if essential, will be the last to be examined in this study.

III. CYCLE ARCHITECTURE AND EFFICIENCY

III.A. Cycle Architecture and Main Parameters

The architecture considered in our study is a modified steam cycle adapted to a Sodium Fast Reactor (SFR) of 650MWe combined with an ORC bottoming cycle (Fig. 1-2). The cycle is composed by the following elements: water pumps, steam turbines (HP and LP), a steam generator, a condenser-boiler, an ORC pump, an ORC turbine and an air cooled condenser. The steam is extracted at the exit of steam generator and at the exit of turbine HP in order to enhance the cycle performance. The cycle efficiency is calculated with THERMOPTIM™ simulation software [18].

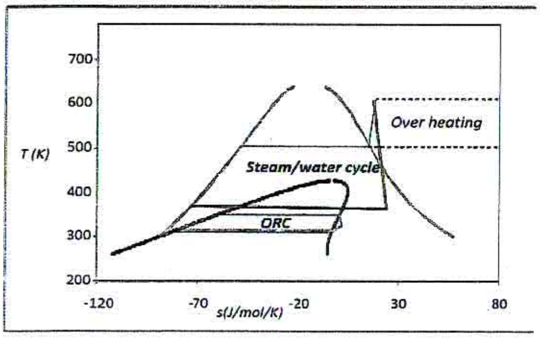

Fig. 1 T-S Diagram of the considered two stage Rankine Cycle

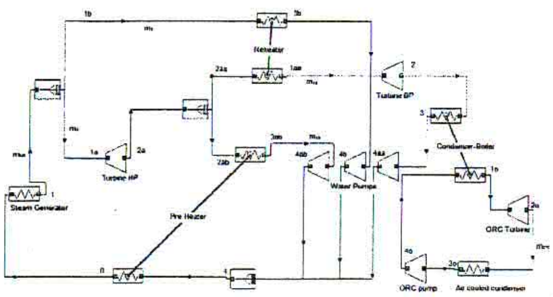

Fig. 2. Structure of Two Stage Rankine Cycle adapted to SFR power plant represented in THERMOPTIM™.

The design of the proposed cycle, although simplified compared to a real cycle, includes in the steam cycle a reheater and a turbine bleeding. Those components make it possible to approach representative efficiency values of the expected real cycle. Indeed, previous studies [19], [20] using an extremely simplified cycles, showed main trends in the ranking of fluids and quantified a weak influence of a regenerator exchanger or a bleed in the ORC cycle.

Pumps and turbines are defined by their isentropic efficiency in this study. The heat exchanges are calculated by considering a constant and fixed pinch temperature in the condenser-reboiler. The pinch temperature in the steam generator and the approach temperature in the air cooled condenser are given. The heat source temperature is maintained constant in the steam generator. The cold source temperature corresponds to the atmospheric dry bulb air temperature.

The STEP is a parameter that we can control in our study. We will analyze the cycle performance as a function of the STEP for the different working fluids.

Table I summarizes the hypothesis made in order to calculate the efficiency of the two-stage cycle.

TABLE I

Components and Operating Conditions for a Two Stage Rankine Cycle adapted to SFR

Cycle component	Variable	Value
Steam generator	Intlet Temperature (T_0)	208 °C
	Outlet Temperature (T_1)	490 °C
	Inlet Pressure (P_0)	230 bar
	Outlet Pressure (P_1)	180 bar
	Mass Flow Rate (m)	680 kg/s
Steam turbine HP	Isentropic efficiency (η_{THP})	0.87
	Expansion ratio	20
Steam turbine LP	Isentropic efficiency (η_{TBP})	0.85
	Expansion ratio	Depending on STEP value
Water pumps	Isentropic efficiency (η_P)	0.56
Pre-heater	Pinch temperature (ΔT_{PH})	4 °C
	Quality of vapor at point 3ab	x=0
Reheater	Superheat (ΔT_{SH})	35 °C
	Mass Flow Rate (m_h)	20 kg/s
Condenser-boiler	Steam Turbine Exhaust Pressure ($STEP$)	Parameter : 0.2 to 1 bar
	Pinch temperature (ΔT_{CB})	5 °C
	ORC boiler temperature (T_{EVAPO})	Depending on STEP value
ORC Turbine	Isentropic efficiency (η_{TO})	0.85
ORC Pump	Isentropic efficiency (η_{PO})	1
Air cooled condenser	Temperature of atmospheric air (T_{air})	11 °C
	Approach temperature (ΔT_{ACC})	29 °C

III.B. Cycle Studies: Results and Discussion

Fig. 3 shows the evolution of the two stage cycle efficiencies for a dry bulb air temperature of 11° C, versus the STEP value and with several working fluids.

It appears that some fluids are less efficient as the value of the STEP increases. When the value of STEP is varied, it means a change in the proportion of mechanical work provided by each cycle. Thus, if the STEP is higher the steam cycle will provide less mechanical work. It is observed that the performance is lower when the fluid has low critical temperature. However, Ammonia which is not an organic fluid provides high efficiencies despite a low critical temperature as compared to other fluids.

The fluids with high critical temperature such as Toluene, Benzene allow very good performance at high STEPs. However both fluids are known for their toxicity. Other fluids such as R1233zd, iC5, R245fa and R365mfc produce acceptable performances. Nevertheless, the two last one have high GWP.

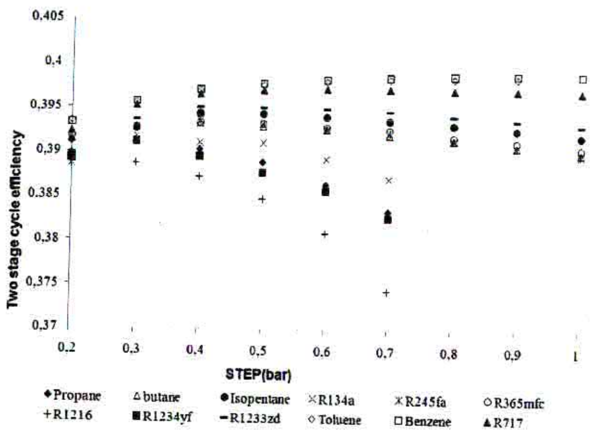

Fig. 3. Two Stage Cycle efficiency calculation results

TABLE II

Fluid Ranking (pure component) according to efficiency

Fluid name	*Tc (K)*	**Efficiency at optimal STEP (dry bulb 11° C)**
benzène	562	0.3989
toluène	592	0.3986
R717 (ammonia)	406	0.3972
R1233zd	438	0.3951
iC5 (isopentane)	460	0.3944
R365mfc	460	0.3933
R245fa	427	0.3933
R600	425	0.3931
R134a	374	0.3918
R290	370	0.3912
R1234yf	368	0.3912
R1216	358	0.3899

It is important to note that the performances at the optimum value are not significantly different regardless of the fluid, but they occur for values of different STEPs. Studies by Liu [19], [20] also have shown that for different dry bulb air temperature values, the ranking of fluids regarding the performances is not affected. Finally this observation led us to refine the selection of media for the realization of the economic study, taking into account the criteria on the choice of fluids (table II). We decided to disqualify Toluene and Benzene because those fluids are too dangerous and we added a R245fa/iC5 mixture whose potential interest would be to limit the high GWP value of R245fa. Fluid properties are evaluated by using Peng-Robinson equation of state[25] and Mathias-Copeman alpha function [26] for thermodynamic properties and Transport Property Prediction method [27-28] for transport properties. Parameters are adjusted on experimental data.

IV ECONOMIC STUDY

In order to quantify and justify the potential cost effectiveness of a two stage Rankine cycle (TSRC), the performance study itself is not sufficient.

We need to determine the impact of fluids on component designs and associated costs. Furthermore, once the designs are decided, the performances in off-design situations of the cycle is calculated over a year period in order to evaluate the potential electric energy that can be released with the cycle. And finally, the calculation of a new levelized cost of energy (LCOE) can be done taking account for local ambient temperature conditions and for typical values of electric power prices over the year.

TABLE III

List of selected fluids including the economic assessment study

Fluid	Full name	Family	Tc (K)	Pc (bar)	GWP	Ashrae classification
R365mfc	1,1,1,3,3-Pentafluorobutane	HFC	460	32.66	890	A2
iC5	Isopentane	HC	460	33.78	11	A3
R1233zd	1-chloro-3,3,3-trifluoropropene	HFO	438	37.72	4	-
R245fa	1,1,1,3,3-pentafluoropropane	HFC	427	36.51	1030	B1
R717	Ammonia	-	405	113.33	1	B1

In addition a mixture of R245fa/iC5 with a mass fraction 0.09/0.91 will be considered in order to keep its GWP <150

IV.A Component Designs

The three main components that need to be considered in a preliminary approach are the two heat exchangers, condenser-boiler (CB) and ACC, and the ORC turbines.

The design point is set for a dry bulb air temperature of 11°C, which corresponds to an average value aver a year in France, and for each of the fluids we use the STEP value corresponding to the optimal efficiency.

The details of the method, including assumptions and correlations, to design the three components and to determine a first assessment of capital cost have been explained in [21]. That work was limited to only 4 of the selected fluids. In the present paper we complete the design assessment for R1233zd and the mixture R245fa/iC5.

The results of the design process can be summarized as follows:

The designs of the two heat exchangers are mainly driven by the thermophysical properties of the fluids, in particular viscosity and conductivity. The surface area of the ACC is one order of magnitude higher than the CB. The heat exchange area for any fluid is about twice the heat exchange required for R717.

As a conclusion of the design study, we decided to disqualify the R365mfc which requires big size components and will not be relevant for economical reasons.

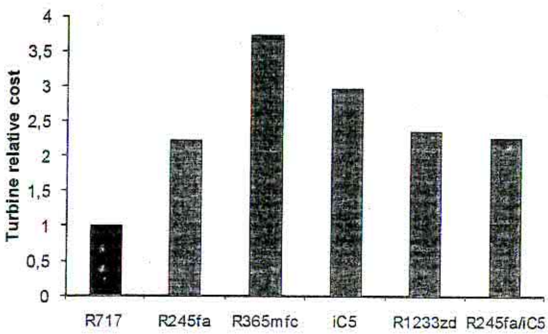

Fig. 4. Turbine relative cost assessment

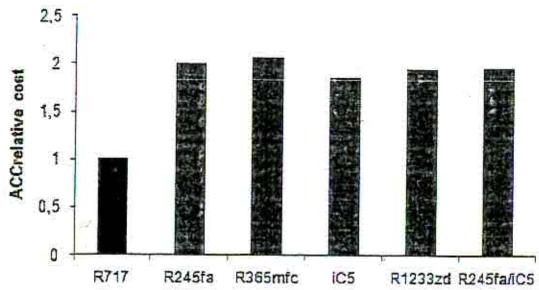

Fig. 5. ACC relative cost assessment

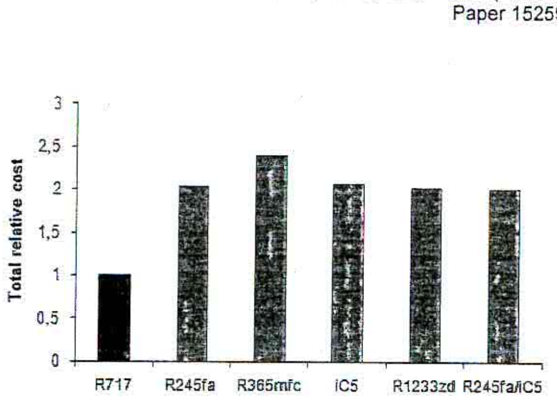

Fig. 6. Total relative cost assessment (sum of components)

The design of the turbines varies mainly with latent heat that determines the mass flow rate, the vapor density and the sound speed that determine a limitation for the last stage of the turbine. The maximum rotation velocity and the number of fluxes are not independent and determine the volumes of the turbines. With a high density and high latent heat, R717 offers the most compact designs since the turbine can operate at 3000 rpm..

In terms of costs, according to correlations reported in Liu, the following Fig.4, Fig 5 and Fig. 6 illustrate the relative costs as compared to R717 for the turbines, the ACC and the sum of the three main components respectively.

IV.B Off design simulation and results

In order to calculate the energy production over one year, we have to determine the efficiency of the two stage Rankine cycle according to different values ambient dry bulb temperature.

To realize that calculation, we derived very simple models for heat exchangers and for the turbines. Concerning the heat exchangers, it appears reasonable to assume that physical phenomena are not much affected by difference in temperature at the boundaries of the components. So we assumed that their global *UA* (ie average heat exchange coefficient *U* time the surface area *A*) is constant. Concerning the turbines, a model has been set that takes into account a relation between the mass flux *m* and the pressures *Pin* and *Pout* at the inlet and outlet respectively and the inlet temperature *Tin*. The relation, as explained in [22] is

$$m = K\sqrt{\frac{P_{in}^2 - P_{out}^2}{T_{in}}} \tag{1}$$

with *K* being the "Stodola" constant that characterizes the turbine.

In addition, it is considered that during period of low air temperature, the expansion of gas within the turbine cannot be completely transferred into valuable mechanical energy. For some fluids, due to their very low density at low pressure, a large part of kinetic energy will be lost.

As a consequence, Fig; 7 illustrates the evolution of efficiency as a function of outside air temperature and fluids. For that purpose, the use of a steam water Rankine cycle with dry cooling was also considered. Fig 7 shows an increase of energy production with low temperature and high density fluids. On the other hand water cycle experiences limitation in power production when the temperature is lower than 5°C.

Thus calculations of the two stage Rankine Cycle with different working fluids were made for typical ambient temperatures near Paris [23]. Table IV reports the comparison of energy production over one year. Depending on the working fluid, an additional power production of 0.8 % to 1.5% can be expected in comparison with dry cooled steam water Rankine cycle. The best score is obtained with ammonia. Other fluids are a slightly less efficient although the R1233zd gives reasonable results.

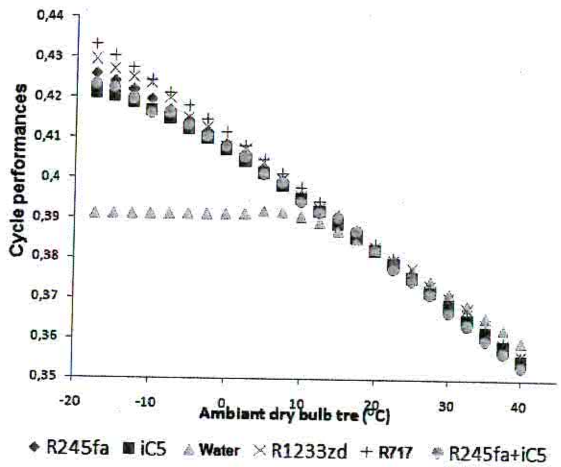

Fig. 7. Cycle efficiencies versus ambient temperature for two stage Rankine cycle and steam water cycle

III.C. Consideration on Cost effectiveness

Impact of the two stage Rankine cycle on LCOE

It is not obvious to make a good quality assessment of cost effectiveness without a detailed engineering work. Moreover, in our study, the difficulty is increased because two of the major components are not available on market (ie : large size ORC turbines and large ACC) but there are no strong reasons to fear that those two components could not be developed. In any cases the costs can vary within an important range of uncertainties (30%).

TABLE IV

Comparison of average energy production over one year using typical temperatures values in Paris area

	Energy production (GWh/an)	Extra production (%)
Steam/water	5626	-
R717	5709	**1.48%**
R245fa	5677	**0,91%**
iC5	5670	**0,78%**
R1233zd	5687	**1.08%**
R245fa/iC5	5676	**0,89%**

In order to presents some preliminary values about the interest of the cycle, the following assumptions were made:

Based on the communication of EDF work with ammonia cycle in 1989, it was stated that, for a PWR of 1300MWe, thanks the reduction in turbine size and associated buildings , the capital cost for a TSRC with ammonia will not be very different) from a steam water Rankine cycle with wet cooling towers.

With the relative cost of components presented in the previous section, it is possible to give a value to the difference in Capital Cost (ΔCAPEX) for the different fluids. The difference of cost is estimated to be of 60 M€.

The extra production (ΔE) of energy has been calculated and the method can be applied for any location, providing that ambient temperature conditions are available.

Difference in operational costs ΔOPEX is assumed to almost 0 M€.

Operation life is 40 years (N) and the discount rate is 8% (r).

Finally, LCOE of the TSRC is expressed as follows

$$LCOE_{TSRC} = \frac{(LCOE_{nuc}\, E_{steam} + \Delta LCOE\ \Delta E\ )}{(E_{steam} + \Delta E)} \qquad (2)$$

With

$$\Delta LCOE = \Delta CAPEX / \sum_{i=1}^{N} \frac{\Delta E}{(1+r)^i} \qquad (3)$$

When applying equation (2), it can be found that LCOE with a TRSC is very close to $LCOE_{nuc}$. $LCOE_{nuc}$ for "operating" nuclear plants in France is about

49.5€/MWh [24]. The preliminary conclusion is that, according to previous assumptions, the difference between the two technical solutions will not be very important on the global cost of electricity production. Nevertheless, some important costs have not been examined and may have an important impact on the economics: cost of buildings, operation and maintenance, safety etc.. so that the conclusion cannot be definitive and additional and more precise studies are needed.

Impact of the two stage Rankine cycle on sales

An important feature of the prices of energy is that they are based on the energy marginal generation cost. As a consequence, in France the price of energy is higher in winter period as the temperatures are lower.

It would be interesting to calculate the benefits that could be generated from an extra production of energy in winter when the prices are higher.

IV. CONCLUSIONS

We investigated in this paper a two stage Rankine cycle for large power plant. This technology could be relevant for the future since it allows operating the plant with dry cooling using a cost effective architecture. The advantages as compared to standard dry cooling are twofold: a large reduction of steam water cycle and an increase of energy production during cold ambient temperatures. This application is slightly different from the classic ORC applications such as waste heat recovery by the fact that we have control over the ORC heat source and the heat exchange with the source is at constant temperature. Several criteria for fluid selection have been listed. 6 working fluids, R245fa, R365mfc, iC5, R1233zd, a mixture of R245fa/iC5 and ammonia have been tested using THERMOPTIM™ simulation software.

The optimal STEPs are evaluated at 11 °C of external air temperature. However, the differences between the cycle efficiencies of the tested fluids are not very significant.

The sizes of turbines have been estimated and designs of main heat exchangers as well leading to assessment of costs for the three main components. Ammonia system seems to be the most compact and the least expensive but due to the toxicity of ammonia, operating with that fluid could be a very difficult issue .For R245fa, R1233zd and iC5 systems are much larger and cost twice more than the ammonia cycle. R365mfc is the least competitive in terms of component size and cost. The R245fa/iC5 is studied because it reduces the GWP negative impact of the R245fa taken as pure fluid.

In order to determine the cost effectiveness of a TSRC, off-design performance of the two-stage cycle has been simulated. Thus, the additional electricity produced at low temperature condition by the two-stage cycle was calculated and used for a preliminary estimation of the LCOE of a plant including the TSRC technology.

The technology seems very promising but need important programs of developments for the technologies of ORC turbines and large ACC. Moreover, thorough engineering studies should be engaged in order to confirm those first values.

NOMENCLATURE

ACC : Air Cooled Condenser

CB : Condenser Boiler

ECS : Energy Conversion System

GWP : Global Warming Potential

LCOE : Levelized Cost Of Energy

ODP : Ozone Depletion Potential

ORC : Organic Rankine Cycle

PWR ; Pressurized Water Reactor

STEP : Steam Turbine Exhaust Pressure

TSRC : Two Stage Rankine Cycle

REFERENCES

1. U. DRESCHER, D. Bruggenann, "Fluid selection for the organic Rankine cycle (ORC) in biomass power and heat plants", *Applied Thermal Engineering,* **27**: 223-229, 2007.

2. U. DESIDERI, G. Bidini, "Study of Possible Optimisation Criteria for Geothermal Power Plants", *Journal of Energy Conversion and Management*, **38**, 1681-1691, 1997.

3. B.F. TCHANCHE, G.r. Lambrios, A. Feangoudakis, G. Papadakis. "Low-grade heat conversion into power using organic Rankine cycles – A review of various applications." *Renewable and Sustainable Energy Reviews,* **15**, 3963-3979, 2010.

4. B. SALEH, G. Kogalbauer, M. Wendland, J. Fischer, "Working fluids for low-temperature organic Rankine cycles", *Energy*, **32**, 1210-1221, 2007.

5. B.T. LIU, K.H. Chien, C.C. Wang, "Effect of working fluids on organic Rankine cycle for waste heat recovery", *Energy,* **29**, 1207-1217, 2004.

6. V. MAIZZA, A. Maizza, "Unconventional working fluids in organic Rankine-cycles for waste energy recovery systems", *Applied Thermal Engineering*, **21**, 381-390, 2001.

7. H. LIU, Y. Shao, J. Li, "A biomass fired micro scale CHP system with ORC", *Biomass and bioenergy*, **25**, 3985-3994, 2011.

8. J.P. ROY, M.K. Mishra, A. Misra, "Performance analysis of an Organic Rankine Cycle with superheating under different heat source temperature conditions", *Applied Energy*, **88**, 2995-3004, 2011.

9. N.B. DESAI, S. Bandyopadhyay, "Process integration of organic Rankine cycle", *Energy,* **34** , 1674–1686, 2009.

10. S. QUOILIN, M. Orsz, H. Hemond, V. Lemort, "Performance and design optimization of a low cost solar organic Rankine cycle for remote power regeneration", *Solar Energy*, **85**, 955-966, 2011.

11. R. CHACARTEGUI, D. Sanchez, J.M. Munoz, "Alternative ORC bottoming cycles for combined cycle power plants", *Applied Energy*, **86**, 2162-2170, 2009.

12. J. FLEURY, C. Bellot, "EDF is testing a 22MWe pilot plant operating with an ammonia bottoming cycle", *ASME Winter Congress*, San Francisco, 1989.

13. J. STALLINGS, F. David, "Summary of the EDFwater ammonia binary cycle research program (CYBIAM) for energy production", *EPRI report 1024685*, 2011

14. ASHRAE, "American standard ANSI/ASHRAE 34. Designation and Safety Classification of Refrigerants", 2010.

15. U. DESIDERI, B Fachini, "A New Gas-Steam-Ammonia Combined Cycle to Reduce Cooling Water Consumption", *Proceedings of the 1992 ASME Cogen Turbo Power*, 193-200, American Society of Mechanical Engineers, New York, USA (1992).

16. United Nations Environment Program, "Handbook for the Montreal Protocol on substances that deplete the ozone layer" (Eighth ed), UNEP (2009).

17. United Nations, "Kyoto Protocol to the United Nations Framework Convention on Climate Change", 1998.

18. R GICQUEL, *THERMOPTIM™ User's manual.* 2010.

19. B. LIU, P. Riviere, C. Coquelet, R. Gicquel, F. David, "Investigation of a two stage Rankine cycle for electric power plants", *Renewable Energy*, **36**, 659-670, 2011.

20. B. LIU, P. Riviere, C. Coquelet, R. Gicquel, F. David, "Potential interest and fluid selection for bottoming cycle applied to nuclear plants", *Proceeding of ICAPP 2013*, Jeju, Korea , april 14-18, 2013

21. B. LIU, P. Riviere, C. Coquelet, R. Gicquel, F. David, "Air cooled two stage Rankine cycle for large power plants operating with different working fluids, performances sizes and costs", *Proceeding of ASME Power conference 2013*, Boston, Massachussets , july 29-august 1, 2013

22. R. GICQUEL, *Systèmes Energétiques Tome 3,* Chapter 15.4, Mines ParisTech, 2009

23. ASHRAE Climate DATA, 2013

24. COURS DES COMPTES, "Les couts de la filière electronucléaire". *Public Report*, 2012.

25. Peng, and Robinson. "A new two parameters Equation of State." Ind. Eng. Chem. Fundam. (15), 1976: 59-64.

26. Mathias, and Copeman. "Extension of the Peng-Robinson Equation of State to Complex Mixtures: Evaluation of the Various Forms of the Local Composition Concept." Fluid Phase Equilibria (13), 1983: 91–108.

27. Huber, Laesecke, and Perkins. "Model for the viscosity and thermal conductivity of refrigerants, including a new correlation for the viscosity of R134a." Industrial & Engineering Chemistry Research (42), 2003: 3163-3178.

28. Huber, and Ely. "Prediction of the thermal conductivity of refrigerants and refrigerant mixtures." Fluid Phase Equilibria (80), 1992: 249-261.

29. Regulation (EU) No 517/2014 of the Europeean Parliament and of the Council of 16 april 2014 on fluorinated greenhouse gases and repealing regulation (EC) No 842/2006